ISSN: 2319-8753

International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 3, Issue 12, December 2014

Vibration Analysis of Isotropic and Orthotropic Turbine Blade Using ANSYS

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ABSTRACT: In this paper focuses on vibration characteristics of turbine blade made up of isotropic and orthotropic (composite) material. The whole research is been carried out in two phases i.e. modeling and modal analysis. Firstly in modeling phase the blade is modeled by using CREO in such as way that parametric analysis can be done and in second phase i.e. analysis the blade is subjected to a cantilever mode and the natural frequency is been computed.Moreover, a comparative analysis between the blade material natural frequencies has been done also with the presence of cutout of different shapes such as circle and rectangle having same volume and mode shapes is generated and reveals the effect of cutout shapes factor in over all stability of turbine blade.

KEYWORDS: Natural Frequency, cutout shape factor, stiffness.

I. INTRODUCTION

During designing a turbine blade various factors are to be considered such as Durability, stability, strength, cost and vibration. The performance of blade structure can be enhanced by diminution of vibration which is advanced technique and most be implemented, which is beneficial in developing better design, high stability and even low cost.

A superior design attitude for diminishing vibration is to split the natural frequencies of the structure from the harmonics of rotor speed which ultimately effects in avoiding resonance where large amplitudes of vibration could severely smash up the structure. For such purpose Frequency placement is one of the ambient techniques.

Fiber reinforced composites and the isotropic materials are in various blade applications has stimulated a considerable amount of study.

The inherent blade vibration grants a prospect to use the blade vibration response for computing any degradation of the rotor blades and thus forecast the onset of blade faults and mitigate the probability of unexpected catalytic failures.

Over the past few decades, various sensors and non-destructive testing (NDT) methods such as fiber optics, laser Doppler vibrometer (LDV), ultrasonic, X-ray, thermal imaging, acoustic emission, strain memory alloy, and eddy current methods [1-3] have been explored for damage detection. Each of the techniques has its own virtues and limitations. In scrupulous, their effectiveness for wind turbine blade condition monitoring and economic viability for such applications is yet to be investigated.

Blades are critical components in the energy conversion of rotary machinery. The quality of the blades' vibratory performance strongly limits the possibility for developing rotary machineries with larger capacities and higher parameterization. In a basic jet engine air is compressor through compressor which is in taken from frontal intake, in compressors by means of these blades the air is compressed due to which air pressure increases and that compressed is further sprayed with fuel and electric spark ignites the mixture. During burning these gases are expands and which pass over another sets of bladed disk which powers the turbine to rotate and produce adequate amount of thrust through nozzle. Jet engines are self sustaining machines as long as fuel is provided they will keep operating.

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II. LITERATURE REVIEW

Pritchard and Adelman [4] originated a mathematical programming optimization model in order to minimize the vibration characteristics by considering minimization of the induced shearing forces at rotor hub. the sizes and locations of the tuning masses along blade span were taken as design variables. The techniques in which vibration modes shapes are altered through mass and stiffness modification in order to make them less them less responsive to the air-loads are termed as modal shaping'' or ''modal tailoring''[5,6].

Rand (1995) conducts and experiment of detecting natural frequency of helicopter blade which is composite thin walled. [7] Since the dependency of the experiment depends upon the angular velocity of rotor and lamination angles of composite blade the test is conducted in vacuum chamber. In test region both symmetric and antisymmetric box-beam specimens were tested and the obtained frequencies had high repeatability and conclude that the natural frequency of blade is a function of lamination modes and their angular velocity.

Sinha and Turner [8] derive a partial differential equation which governs the motion for the transverse deflection of a rotating pre-twisted plate. The warping effect of twist-bend in cross section has been included in Strain–displacement relationships and the blade is subjected to cantilever boundary condition with quasi-static load act on it due to centrifugal force. Rayleigh-ritz technique is been used to solve the governing equation of motion and transformed into a matrix-eigenvalue. The obtained result is well validated with literature test data both for thin and thick plate.

Maalawi et .al 2002 develop a new model for design wind turbine (Horizontal axis). In their model several optimization variables are considered which enhance the performance of the turbine such as length of each element, cross section area and radius of gyration. The model is examined under the problem is formulated as a non-linear mathematical programming problem solved by multi-dimensional search techniques.[9] Floquet's transition matrix theory is used to evaluate aero elastic stability boundaries and steady-state response and conclude that the obtained results are efficient and produces improved designs as compared with a reference or baseline design.

Kuang and Hsu 2002 applied (DQM) differential quadrature method for tapered pre-twisted orthotropic composite blade to analyzed vibration characteristics. And compares with Euler–Bernoulli beam model. Some other coefficients are also evaluated such as linear external damping and Kelvin–Voigt internal along with the inclination angle rotational speed, fibre orientation and internal- external damping on the natural frequency for composite blade. [10]

Forbes and Randall 2013 use non-contact method for determining rotor blade natural frequency in casing casing vibration measurement. It is found that the effect of pressure on casing due to vibration casing pressure increase and analytical model is also been used to analyzed pressure signal due to vibration which includes measureable information. [11]

Wang et. al 2014 proposed a method for examining blade damage and its diagnosis. FEM tool is implemented for dynamics analysis of (MSDC) mode shape difference curvature which is adopted for damage detection /diagnosis and modal response investigation. They conclude that the relationship between modal constraints and damage information related to location and size is very obscured particularly for larger size blades. Furthermore, structure and dynamic nature for larger size blades are different in comparison with smaller sized blades.[12]

Ashwani et. al 2014 introduces a new material AL2024 for wind turbine blades. With the means of FEM suitability, analysis of Complex objects and geometries have been carried out. The natural frequency of modeled wind turbine blade is evaluated by using ANSYS 14.0 and the blade is modeled with the help of Solid Edge software. The obtained result shows the good verification with the experimental result.

Poursaeidi et. al 2012 present an experimental and computational verification of effects of natural frequencies on the failure of compressor blades. The analysis is been carried out by using FEA software platform ANSYS and Fractography analysis is done on fractured surface of blades.[15] The blade is subjected under static and dynamic forces in ANSYS and revealed that the first and second natural frequency modes are the main reason for the fatigue fractures of blades.

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III. MATHEMATICAL MODELLING

The purpose of modal analysis is to calculate natural frequencies and vibration modes, and this problem comes down to eigen value and eigen vector problem. The FEM basic equation of dynamics problem for non damping free vibration is:

$$M\ddot{\delta}(t) + C\dot{\delta}(t) + K\delta(t) = P_f(t) \tag{1}$$

Let the damping force and the excitation force in eq. 1 is zero:

$$M\ddot{\delta}(t) + K\delta(t) = 0 \tag{2}$$

Free vibration of any elastic body can be resolved into a superposition of a set of harmonic vibrations. Let the harmonic vibrations solution of eq. 2 is:

$$\delta(t) = \delta_0 \sin \varpi t \tag{3}$$

Substituting eq.3 into the basic equation of free vibration, we can get eq. 4:

$$(K - \varpi^2 M)\delta_0 = 0 \tag{4}$$

Since amplitudes of nodes in structure δ_0 are not all zero, value of coefficients determinant must be zero:

$$\left|(K - \varpi^2 M)\right| = 0 \tag{5}$$

Since the stiffness matrix K mass matrix M is n order matrix, eq.4.3.5 is a n order equation of ϖ^2. The natural frequencies are generalized eigenvalue and can be solved by Rayleigh quotient method:

$$\varpi_i = \frac{\delta_{0i}^T K \delta_{oi}}{\delta_{0i}^T M \delta_{oi}} \tag{6}$$

Natural frequencies:

$$f_i = \frac{\varpi_i}{2\pi} \tag{7}$$

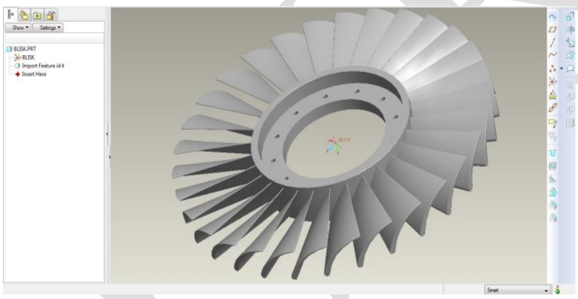

Figure 1 Modeled Geometry

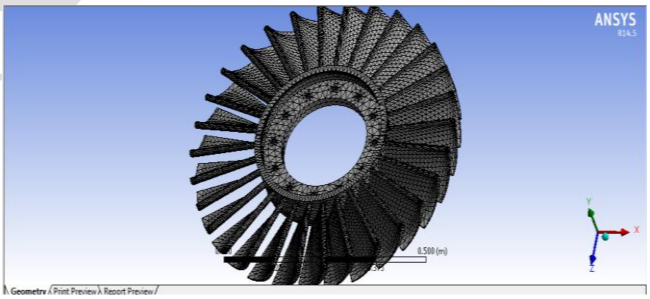

Figure 2 Mesh model

DOI: 10.15680/IJIRSET.2014.0312069

ISSN: 2319-8753

International Journal of Innovative Research in Science, Engineering and Technology

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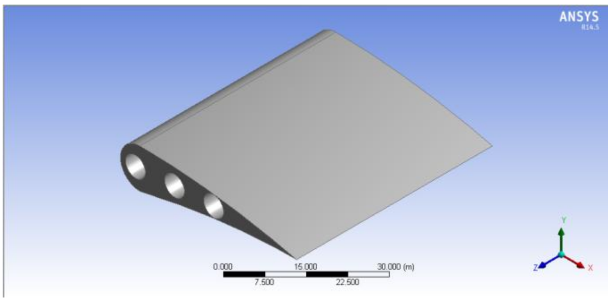
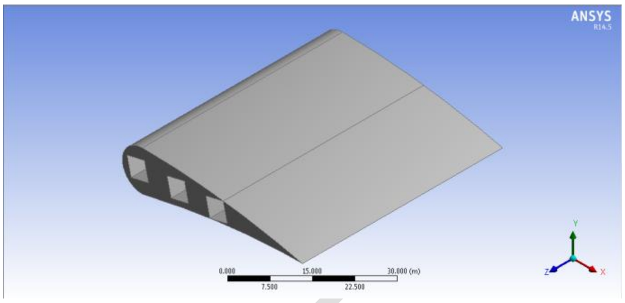


Circular Cutout



Square cutout

Figure 3 model Blade Geometry with different cutout configuration

Figure 3 shows different modes of vibration occurred in CCFF boundary condition.

Table 1: Material properties

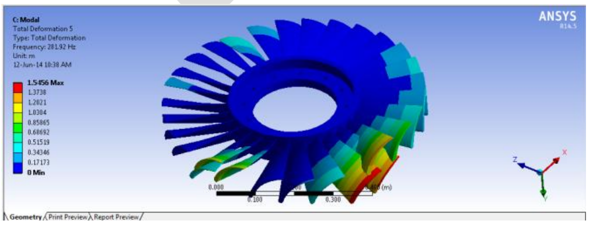
S.no	Material	Density, ρ	Modulus of Elasticity, E	Poisson's ratio, υ
1.	Structural Steel	7850	2e11	0.3
2.	Stain less steel	7750	7.93e11	0.31
3.	Titanium Alloy	4620	9.6e11	0.36
4.	Aluminum Alloy	2770	7.7e10	0.33
5	T-Graphite Epoxy	1600	1.85e11,10.5e9	0.28
6	B-Boron Epoxy	2000	2.08e11,1.89e10	0.23

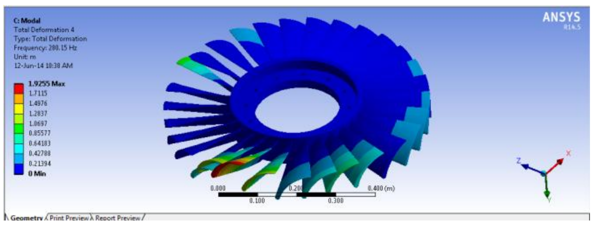
Table 1 show the material properties of blade which are used during the analysis .Material properties like Density , Modulus of Elasticity and the Poisson's ratio.

Table 2 Validation of Natural frequency of turbine blade

Natural frequencies obtained by FEM and modal test for an blade			
Mode no.	**FEM Ref. [12]**	**Expt. Ref. [12]**	**Present ANSYS**
1	23.402	22.8	23.2501
2	80.009	89.4	80.0125
3	94.249	92.2	94.461
4	222.01	192.1	221.982
5	333.07	346.8	333.002

table 2 shows the validation of natural frequency of blade the obtained result shows the good agreement with the experimental and computation result of stated literature with different condition.





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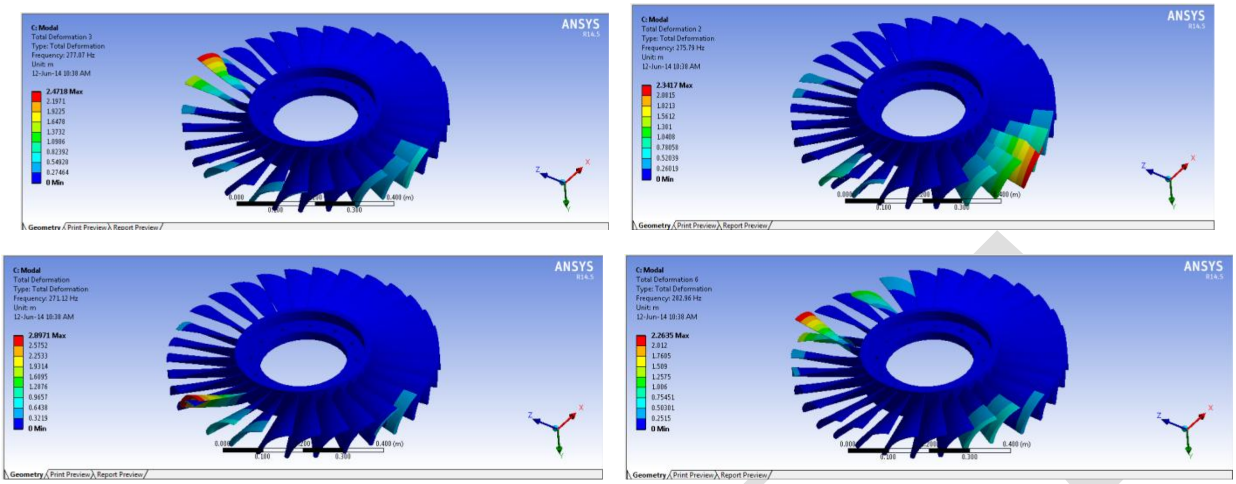
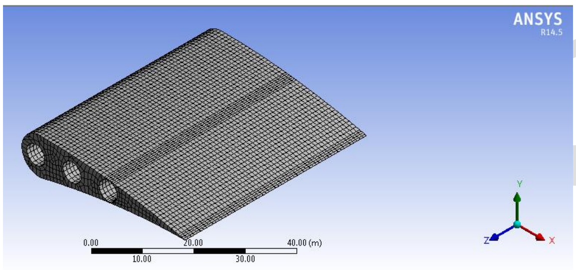
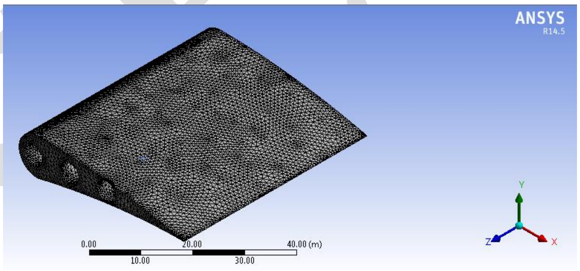


Figure 4 Modes of turbine blade rotor at different frequency



Mapped Meshing



Refine meshing

Figure 4 Mesh configurations for circular cutout i.e. Mapped and Refine Meshing

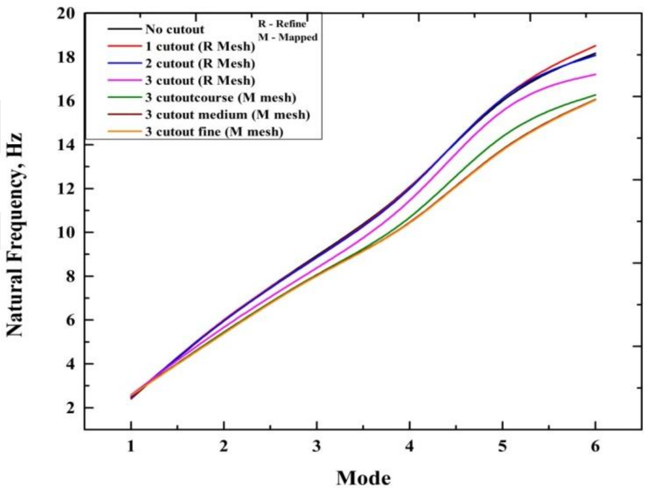


Figure 5 Variation of Natural Frequency of Blade for varying number circular of cutout with different Mesh Configuration

IJIRSET

ISSN: 2319-8753

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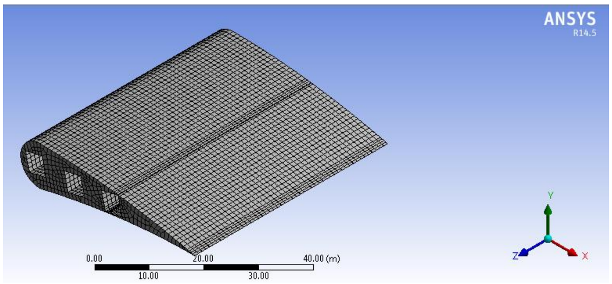
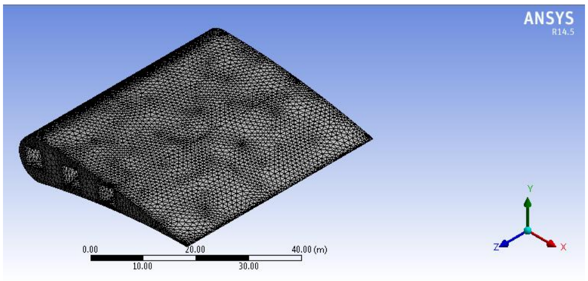
Figure 4-5 shows the variation of Natural Frequency of Blade for varying number circular of cutout with different Mesh Configuration. It has observed that on increasing number of cutouts the natural frequency increases. Moreover, on comparing mesh configuration for optimum frequency for 3 cutouts is found to be maximum in refine meshing as compared to mapped meshing but in refine meshing the number of elements are more in comparison with mapped meshing. However in mapped meshing the mesh sizing element for three mapped meshing i.e. course, medium and fine, the sizing element is remain same, which ultimately results in decrease in natural frequency.



Mapped Meshing



Refine meshing

Figure 6 Mesh configurations for Square cutout i.e. Mapped and Refine Meshing

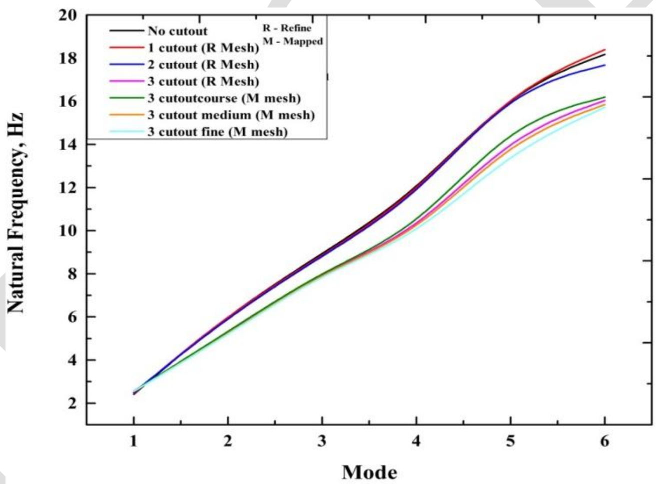


Figure 7 Variation of Natural Frequency of Blade for Varying number Square of cutout with different Mesh Configuration

Figure 6-7 shows the variation of Natural Frequency of Blade for varying number square of cutout with different Mesh Configuration. It has observed that on increasing number of cutouts the natural frequency increases. Moreover, on comparing mesh configuration for optimum frequency for 3 cutouts is found to be highest in refine meshing as compared to mapped meshing but in refine meshing the number of elements are more in comparison with mapped meshing. However in mapped meshing the mesh sizing element for three mapped meshing i.e. course, medium and fine, the sizing element is remain same, which ultimately results in decrease in natural frequency

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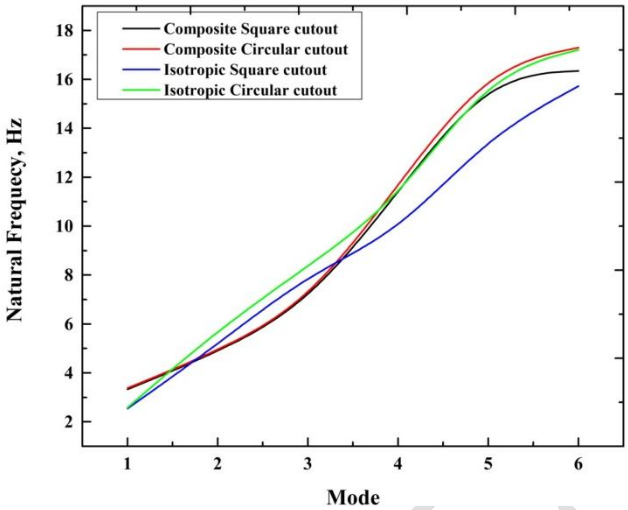

Figure 8 Comparison of Natural frequency of blade with different cutout with different material

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Here it can also be revealed that the in every mesh configuration the natural frequency for circular cutout is always slightly greater in comparison with square cutout even though have same volume of cutout have been placed at same equal-distance between the from each other. The significance behind such a behavior is due to mass, stiffness and boundary condition to which the blade is subjected to i.e. CFFF.

The modal parameters such as natural frequency, mode shapes and modal damping are all functions of structural physical properties i.e. mass, stiffness and damping. If any form of singularities such crack, cutout, holes etc the stiffness will reduced. Therefore, this ultimately decreases the natural frequency and changes the mode shape of the structure.

Figure 8 shows the Comparison of Natural frequency of blade with different cutout with different material. It has been observed that the natural frequency of composite material is more in comparison with isotropic material. The significance for such a behavior is due to variation in mass which is function of density along with this the physical shape factor of structure also responsible for such change. And due to which the stiffness of structure get affected.

IV. Conclusion

It can also be conclude that the natural frequency of an composite material is always more as compared with isotropic material. Presence of cutout affects the frequencies in both boundary conditions Increasing Cutout diameter affects the frequency.

Modal parameters (natural frequencies, mode shapes and modal damping) are functions of structures physical properties (mass, stiffness and damping).If any cutout exists in a structure, the stiffness will be reduced. The reduction in stiffness may cause a decrease in the natural frequency and change the mode shape of the structure.
On changing mesh configuration the natural frequency of blade changes.

Increasing the number of cutout significant increase in frequency has been found. Using different material the frequency of vibration is quite different this is due to varying in density of modulus of elasticity as they are function of flexure rigidity.

The natural frequency in case of isotropic and orthotropic and composite blade is Dependent on the aspect ratio. Since the blade is subjected to CFFF boundary condition but if ton changing blade orientation the mode of frequency changes.

IJIRSET

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